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# CURTISS-WRIGHT'S EXPERIMENTAL ROTATING COMBUSTION ENGINES

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# CURTISS-WRIGHT'S EXPERIMENTAL ROTATING COMBUSTION ENGINES

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## HISTORICAL SURVEY

ALMOST AS SOON as inventors and engineers made a successful steam engine in the 18th Century, they were also working on rotary engines. However, the reciprocating principle was more obvious and posed less problems for the operating cycle and the sealing of high pressure and was, therefore, predominant for almost two centuries. The first applications of the rotary principle can be traced back to James Watt and his co-worker, Murdock, the latter having reportedly made a steam expansion engine similar to an ordinary gear pump, but with wooden bars as sealing elements. Murdock's engine delivered half a horsepower and was used by him to drive a lathe in his workshop. Other famous engineers worked on rotary engines, for example, John Ericsson, the builder of the first armored steel vessel, the "Monitor"; Westinghouse; de Laval; and Parsons. None of their designs, however, were successful. Other rotary machines, such as compressors and pumps, have less stringent requirements for the cycle and the seals than engines. Therefore, quite a number of designs using the rotary principles were put into practice and became commercial successes. That many people were also working on rotary engines is best shown by the fact that hundreds of patents were granted which display a multitude of configurations and designs.

A number of patents to an American inventor, J. F. Cooley (and the Cooley Epicycloidal Engine Co.) in the first decade of this century, and patents to a French inventor, Sensaud de Lavaud, shortly before World War II, show interesting examples of such engines. However, no reliable test data were ever made available. This can only mean one of two things:

1. The designs were not even capable of short test runs.
2. The test results were so disappointing that publishing them was not warranted.

An English book on the history of rotary engines and pumps, published in 1939, gave no hope for a successful realization of a rotary engine.

In 1926, the German engineer, Felix Wankel, showed interest in rotary engines and started to study and investigate them systematically.

He established an institute where he also designed and built test rigs for the evaluation of sealing elements. As an intermediate result of his research on seals, he developed a sealing arrangement for a rotary disc valve for a reciprocating engine which was successfully operated and tested as a liquid-cooled high performance aircraft engine.

In 1951, he was able to reopen his Research Institute with the financial assistance of three German concerns. He then investigated known and new rotary machines. This study led him to the conclusion that three main problems had prevented the rotating engine from achieving success so far:

1. The multiplicity of possibilities of arrangements and cycles for rotary engines which almost overwhelmed the inventors.
2. The problem of sealing high pressure chambers. This involved not the simple task of sealing a circular cylinder (like the piston ring in reciprocating engines), but the more complex problem of sealing in several planes, with the sealing of corners as the most difficult.
3. The problem of developing a proper thermodynamic and gas cycle with adequate port areas and timing of events.

## WANKEL ENGINE PRINCIPLE

The combination of a suitable geometric and kinematic arrangement which offered good sealing possibilities, and could incorporate an efficient cycle within one engine, had to be found. Wankel eventually arrived at two rotating combustion engine configurations which appeared superior and were subsequently built and operated satisfactorily.

In both configurations, the engine has two main components, an outer rotor having a cavity within which an inner rotor is relatively rotatable. In practice, the outer rotor preferably consists of a center housing and two attached parallel side housings. The two rotors have spaced but parallel axes. The inner surface of the center housing is an epitrochoid while the outer surface of the inner rotor approximates the inner envelope of this epitrochoid during their relative rotation. The apexes of the inner rotor are in contact with the epitrochoid at all times which

simplifies their sealing. The two rotors are rotated in the same direction about their respective axis at a speed ratio of the outer rotor to the inner rotor of n + 1/n when n is equal to the number of lobes of the epitrochoid.[1] The speed ratio particularly suitable for an efficient and still simple cycle is 3:2. This geometry is shown in *Figure 1*.

Two positions of the rotor are depicted which give the maximum volume of a working chamber (at bdc of this chamber) and the minimum volume (at tdc). The difference between these volumes, of course, constitutes the displacement of the engine and their ratio the theoretical volume or compression ratio. The volume change with rotation is sinusoidal. A cutout in the inner rotor provides the actual desired compression ratio and a gas transfer passage at tdc. In one configuration, both rotors rotate at this speed ratio while in the other configuration, the outer rotor is stationary and the inner rotor has a planetary motion about the axis of the outer rotor. The relative motion of the two rotors is, however, the same in both cases and will be demonstrated later on. The configuration in which both rotors rotate (which will be referred to as the "dual rotation" type) has advantages from the standpoint of the seal operation and since Wankel was very much concerned with the sealing problem, his first engine was based on this configuration. It was designed by him and his associate, engineer Ernst Hoeppner, for a displacement of 125 cc (7.6 cu. in.).

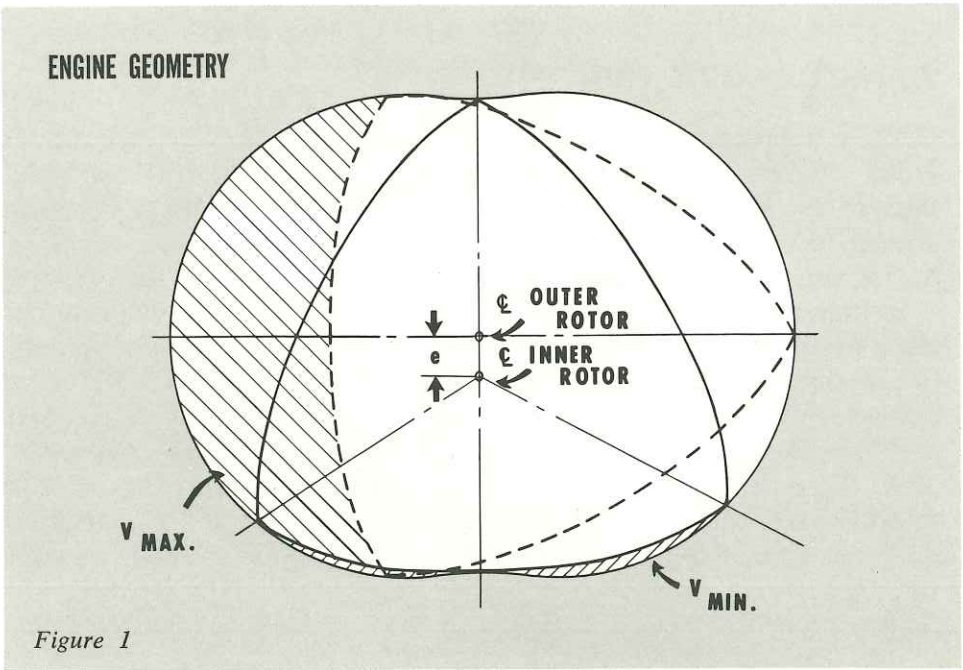



*Figure 1*

[1] S. de Lavaud had proposed an engine in which the outer rotor turns slower than the inner rotor. Such a configuration cannot successfully fulfill the last two of the above three requirements.

## NSU-WANKEL EXPERIMENTAL ENGINES

The longitudinal section of this engine *Figure 2* depicts the outer rotor and the inner rotor whose parallel axes are at a distance e and which rotate in the same direction with a speed ratio of 3 : 2. The engine, therefore, is fully balanced and does not have any mechanical vibration. The outer and inner rotor are connected by gears which guarantee exact positioning of the two rotors but do not transmit the engine torque. The air fuel mixture enters the chambers through the carburetor, the hollow shaft of the inner rotor, intake ports in the inner rotor and cutouts in the side housings of the outer rotor. The inner rotor contains three spark plugs, one for each chamber, which are supplied by a breaker shown on the right. Both the inner and the outer rotor are supported by needle bearings. The outer rotor and, at a later stage of development, the inner rotor as well, are water cooled. This dual rotation engine and its gear arrangement and water cooling make for a rather complex design and intake system but it served its purpose in supplying basic information on the thermodynamic and gas cycle and the performance of the engine.

**LONGITUDINAL SECTION OF FIRST EXPERIMENTAL RC ENGINE WITH DUAL ROTATION**

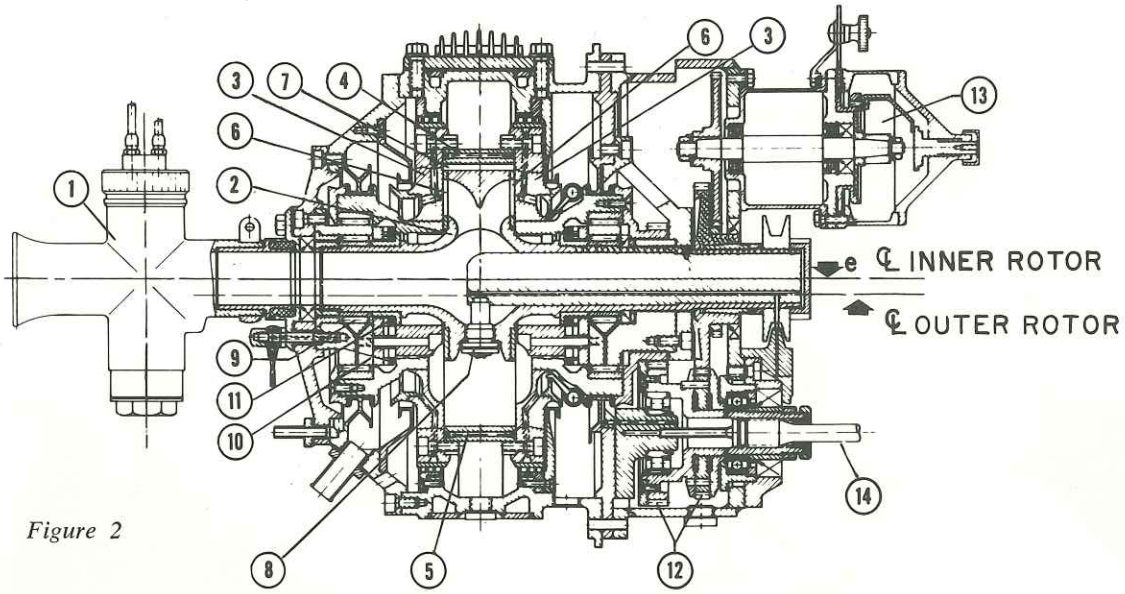



*Figure 2*

1. Carburetor.
2. Inner Rotor.
3. Outer Rotor—Side Housings.
4. Outer Rotor—Center Housing.
5. Cooling Channels.
6. Sealing Side Plates.
7. Apex Seals.
8. Spark Plug.
9. Bearing for Inner Rotor.
10. Bearing for Outer Rotor.
11. Low Pressure Gas Seals.
12. Gears for Outer and Inner Rotor.
13. Breaker.
14. Output Shaft.

*Figure 3* shows the actual engine partly disassembled, the inner rotor, one of its spark plugs and intake ports, the outer rotor with the cutouts in the side housing, and the trochoidal shape of the center housing, the cooling channels in the center housing, and the stationary engine housing with exhaust outlets. The engine exhaust emerging from the single exhaust port in the outer rotor is collected in an annular passage between the outer rotor and the engine housing and leaves through these stationary outlets.

*Figure 4* shows the inner rotor partly disassembled, the sealing plates with sealing strips, the apex seals which are made in five pieces for good sealing of the corners, and the pin which provides sealing between the sealing strips, the apex seals, and the side housings of the outer rotor.

Under Dr. W. Froede's direction, this engine was built and tested by NSU[2] and eventually developed 28.4 hp at 17,000 rpm of the outer rotor (11,300 rpm of the inner rotor) which is equivalent to 1.23 hp per cu. in. displacement.

A single rotation engine of the same displacement was subsequently designed. It was much smaller and simpler than the dual rotation engine although its sealing elements were essentially the same.

At this point in the development, Curtiss-Wright became interested in this type of engine because it showed promising capabilities with respect to simplicity and costs, complete freedom of mechanical vibration, small bulk and weight, elimination of speed restricting valve gear mechanisms, and an efficient operating cycle. Curtiss-Wright concluded a license and engineering assistance agreement with NSU and Wankel[3] which covers the full scope of sizes and configurations of rotating combustion engines. The company concentrated first on a bigger engine, eight times the size of the two German experimental engines, and has designed, built and tested a great number of experimental engines of the single rotation type with 60 cu. in. displacement.

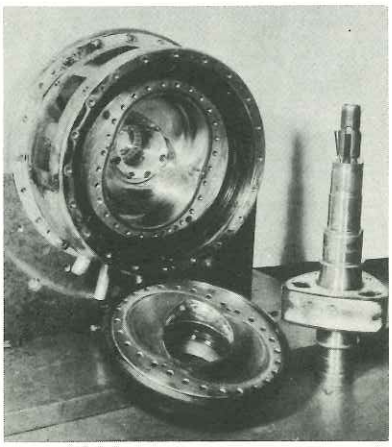


*Figure 3*
*First Experimental Engine, Partly Disassembled*

*Figure 4*
*Inner Rotor, Partly Disassembled*

[2] NSU Motorenwerke Aktiengesellschaft, Neckarsulm, W. Germany.

[3] Wankel G.M.B.H., Lindau/Bodensee, W. Germany.

## CURTISS-WRIGHT ENGINES

**EXPERIMENTAL ENGINE RC6:** The general design of this engine is illustrated in *Figures 5 and 6; Figure 7* shows the components of one of the initial experimental engines. *Figure 8* shows the engine assembled and *Figures 9-13* show the cycle sequence:

1. Fuel-Air Intake.
2. Fuel-Air Compression.
3. Fuel-Air Ignition and Combustion.
4. Power Phase.
5. Exhaust.

**RC6—LONGITUDINAL AND CROSS SECTIONS**

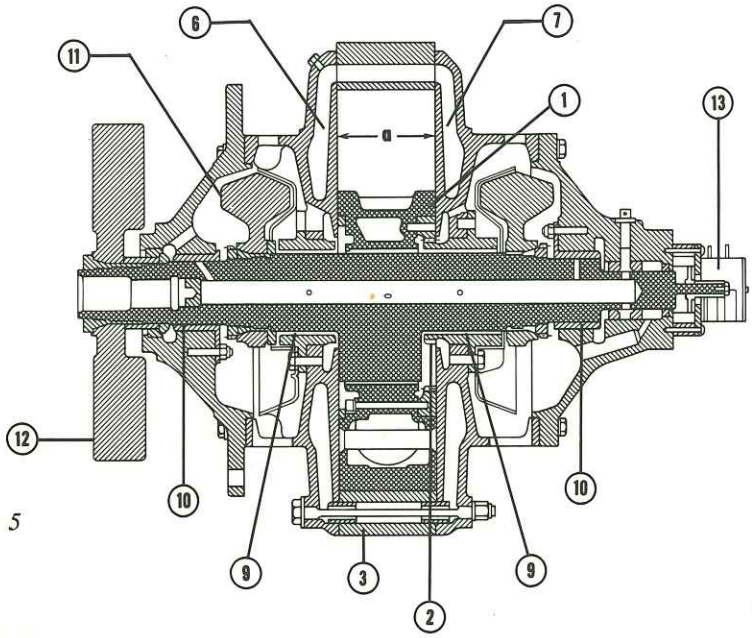



*Figure 5*

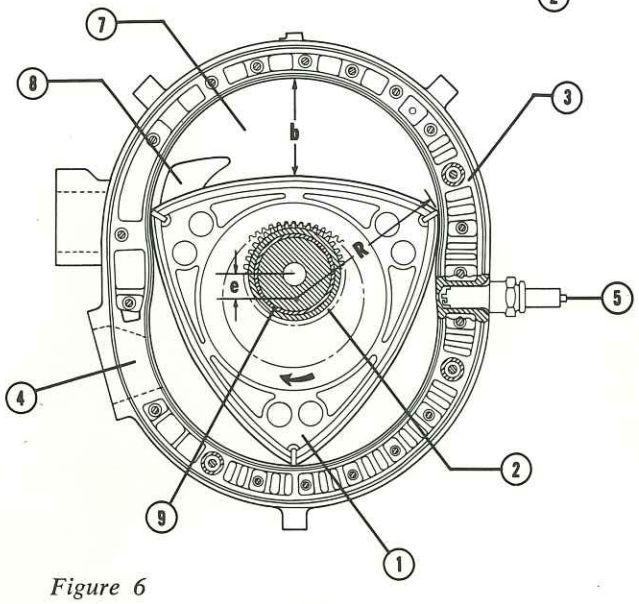



*Figure 6*

1. ROTOR WITH INTERNAL ROTOR GEAR
2. STATIONARY GEAR
3. ROTOR HOUSING
4. EXHAUST PORT
5. SPARK PLUG
6. SIDE HOUSING — DRIVE SIDE
7. SIDE HOUSING — ANTI-DRIVE SIDE
8. INTAKE PORT
9. MAIN BEARING (INNER)
10. MAIN BEARING (OUTER)
11. BALANCE WEIGHT
12. FLYWHEEL
13. IGNITION CONTACT MAKER

a = AXIAL WIDTH OF CHAMBER
R = GENERATING RADIUS
e = ECCENTRICITY
k = R/e
b = MAXIMUM BREADTH OF CHAMBER

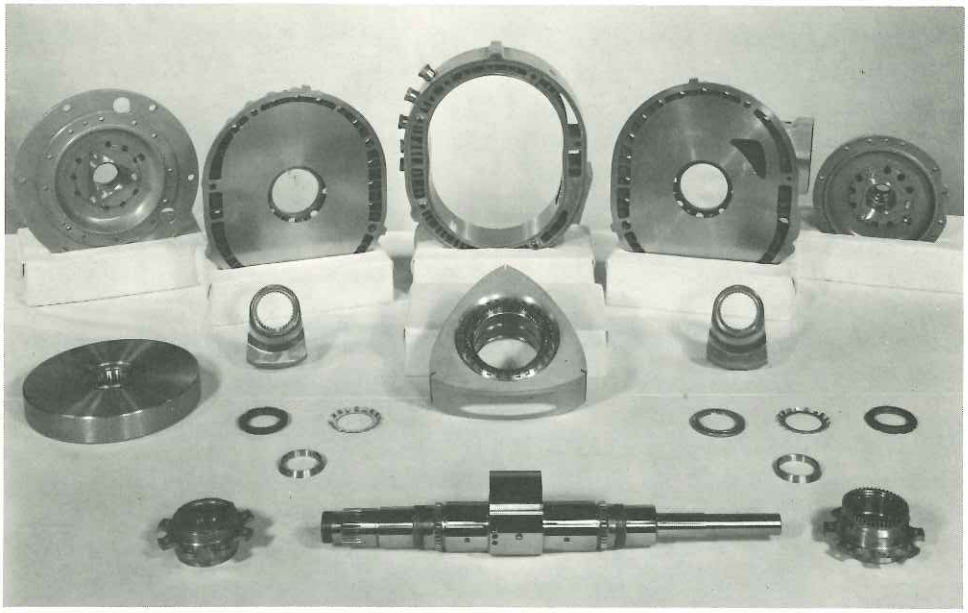


*Figure 7 — Components of First Curtiss-Wright Rotating Combustion Experimental Engine*

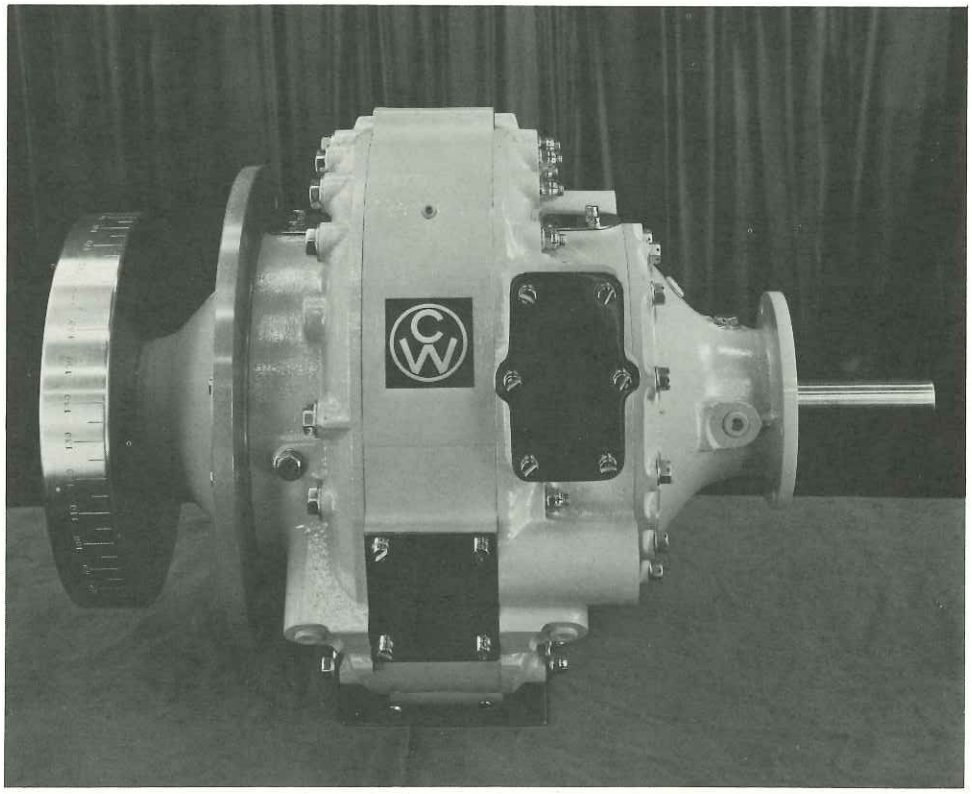



*Figure 8 — Experimental Engine Assembled*

*Figure 9 — Fuel-air Intake*

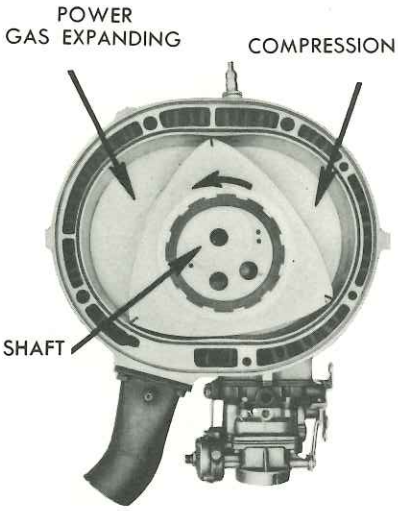



*Figure 10 — Fuel-air Compression*

*Figure 11 — Fuel-air Ignition*

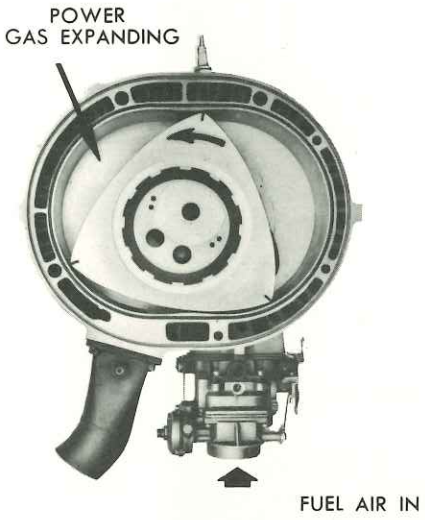



*Figure 12*
*Power Stroke – Gas Expanding – Intake Opening – Exhaust Closing*

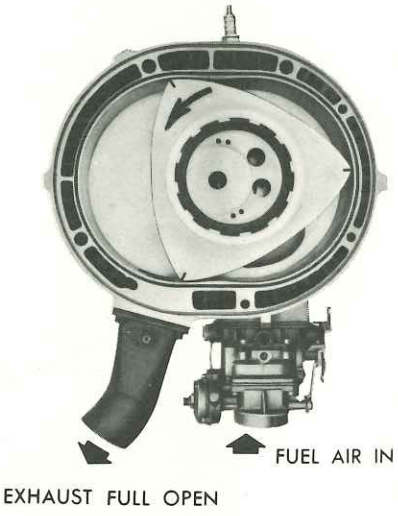



*Figure 13 — Exhaust*

*Figure 14* compares the cycle of a single unit rotating combustion engine with a single cylinder 4-stroke cycle reciprocating engine. Plotted are chamber pressures versus crank angle covering four revolutions of the crankshaft for each engine. The diagram shows that during these four revolutions, the reciprocating engine, of course, has two power sequences, whereas the rotating combustion engine has four. This being significant in itself, the diagram also shows that each power sequence of the rotating combustion engine covers a wider crank angle than reciprocating engine, the ratio being 3/2.

Consistent with this are, also, the intake and exhaust flow, both of which have shorter dwell periods for the rotating combustion engine, thus reducing intake and exhaust losses and providing a smoother operation for single and multi-rotor engines. Curtiss-Wright designated this engine as the "Rotating Combustion Engine."

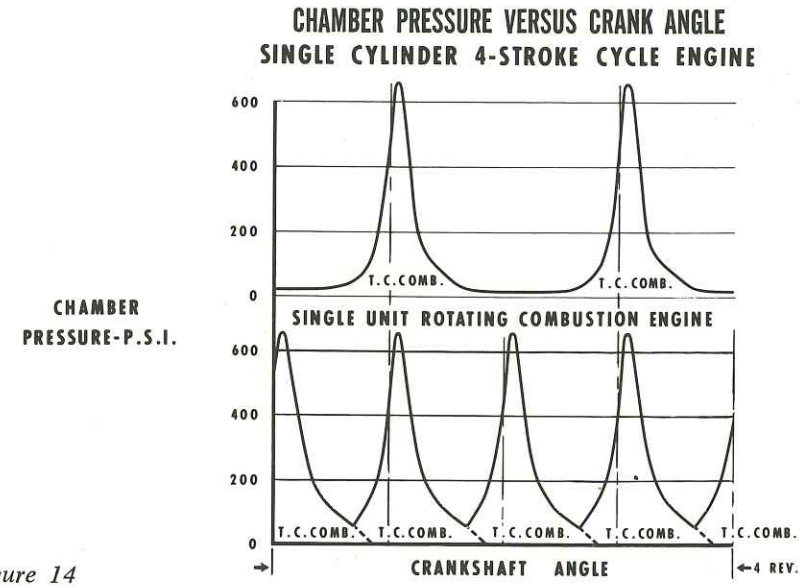



*Figure 14*

GENERAL DESIGN INVESTIGATIONS – In order to arrive at the optimum design of the power section, as shown in *Figures 5 and 6,* we investigated the effects of various geometric variables such as the K factor ($K = R/e$) which represents the ratio of the generating radius R of the trochoid to its eccentricity e, the ratio of the axial width of the chamber a and its breadth b at its maximum volume, and different

porting configurations. The effects of the K factor on the displacement of the engine, the areas and timing for side ports, the overlap, and the combustion chamber shape are rather complex and cannot be expressed in simple formulas mainly due to design considerations, e.g., the accommodation of gas and oil seals. The general relationships have been amply described (1 through 5)[4], and our own conclusions led to the selection of a single K factor for an experimental gasoline engine which allowed us to accommodate for the air intake a single side port, dual side ports, and peripheral ports within one engine size, with a minimum modification to the important components. A specific design for either a dual side port or a peripheral port would have permitted a greater axial length of the engine, with a resulting reduction of R and the "diameter," and an increase of its speed for the same apex velocity, but other considerations, such as mechanical loads and shaft stresses, made it advisable to postpone these investigations until we had a better understanding of the basic engine.

A diesel version will necessitate a higher K factor in order to attain a higher compression ratio.

The mechanical design of the experimental engines is conservative without special efforts toward a low weight. The crankshaft runs in two main sleeve bearings adjacent to the eccentric, and in two more bearings outside the balance weights. The flywheel is separately mounted and easily changed for the investigation of its effect on the engine operation and torsional vibration characteristics.

COOLING – Liquid cooling has been used for both the housing and the rotor. The former is water cooled and the latter is cooled by lubricating oil pumped through the main shaft.

1. Housings. The rotating combustion engine, unlike a conventional reciprocating engine, always exposes the same stationary housing locations to the same portion of the cycle; the variation of heat input per unit of gas-exposed area along the trochoid periphery is shown in *Figure 15*. To match this distribution, the housing cooling was designed as a multipass forced-flow system, as schematically illustrated in *Figure 16*. The cooling liquid flows back and forth through the rotor housing and between the end housing walls. The rotor housing flow passages are parallel to the engine axis; the internal ribbing of the end housings redirects the flow, as shown in *Figure 17,* where the trochoidal rotor housing is in position and the wear face of the end housing has been removed.

The flow area and wetted perimeter, and thus the hydraulic diameter, vary from pass to pass, and are adjusted to match the combustion cycle

[4] Bibliography at end of brochure.

heat input variation. For example, in hot regions of the engine, the flow velocity is high; the amount of heat transfer surface, accomplished by additional ribbing, is large; and the ratio of the total flow area to the total wetted perimeter per pass is small. This series forced-flow configuration permits balanced cooling with large, clog-free, easy to manufacture passages, and avoids vapor accumulation. Furthermore, the large passages permit a cooled wet-bolt application.

This system minimizes temperature variations of the housings. Thermal distortions are further restrained because of the type of structure employed: the outer walls of the housings run cold and maintain the designed shapes while permitting uniformly thin heat transfer walls. The locations of maximum heat input correspond to the locations of maximum stress, so that the addition of ribs, to improve heat transfer, serves to accommodate structural needs as well.

The rotor and side housings are located by dowels and clamped together by numerous bolts which are spaced consistent with the gas loads and located within the cooling passages and hollow dowels.

2. Rotor. A rotor made from an aluminum forging offers the benefits of lightweight and high thermal conductivity, but requires adequate cooling.

One of the most successful aluminum rotor designs is cooled by forced circulation of oil from the engine lubrication system. The cooling circuit is schematically illustrated in *Figure 18*. Oil is supplied from the hollow shaft, through radial feed holes in the eccentric, and then to the rotor bearing central annulus. The oil then divides, one part to lubricate the bearing and the other to cool the rotor.

Integral fins in each rotor lobe provide adequate cooling surface and a small hydraulic diameter. A recess at each end of these finned passes acts as a manifold for distribution of the oil.

The oil from the rotor exit, after passing the bearing end annulus, is discharged from timed shaft eccentric passages directly into the housing drain annulus.

The oil entering temperature, temperature rise, and period of contact with the metal are controlled to avoid coking or other lubricant deterioration. To date, we have had no evidence of adverse effects in this regard. While the oil flow of this design is continuous, the velocity varies as a function of the system dynamics: shaft and rotor inertia pumping superpose a roughly sinusoidal pressure increment onto the oil supply pressure. The cooling oil is metered at the exit; in this way, the rotor cavities are maintained full. Cast iron rotors can be made very much simpler than aluminum rotors but at present, not quite as light. Their cooling system is, however, different; it is adjusted to the lower thermal conductivity of cast iron and the thinner rotor walls. It is expected that by further design and casting improvements, the weight of iron rotors will approach that of aluminum rotors.

PLOT OF HEAT INPUT PER UNIT OF GAS-EXPOSED AREA

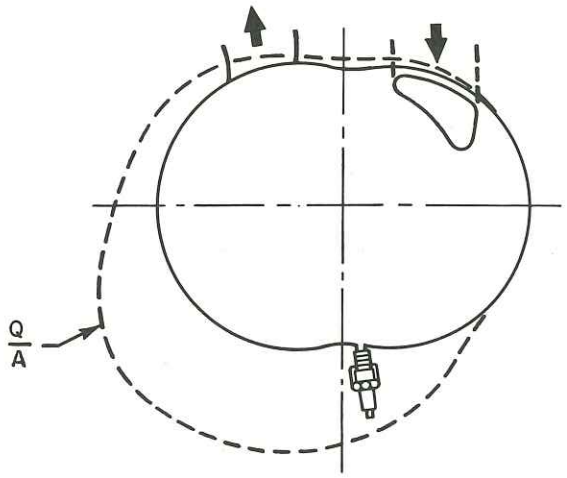



*Figure 15*

SCHEMATIC OF HOUSING COOLING SYSTEM

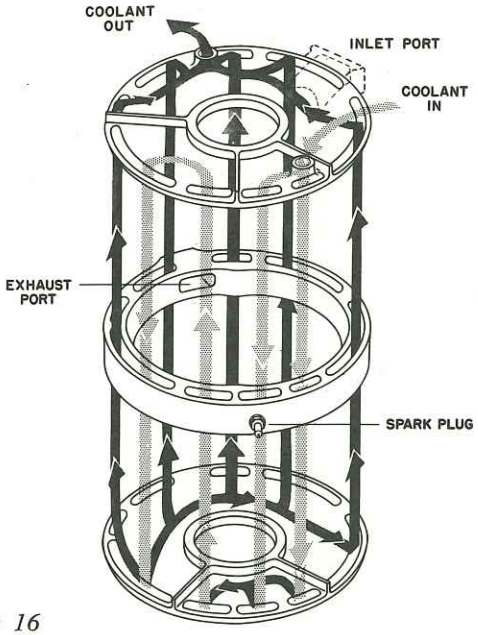



*Figure 16*

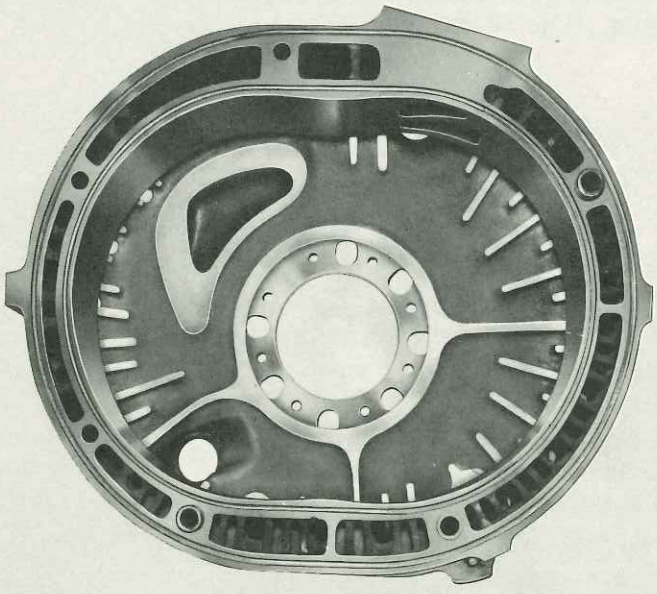


*Figure 17 — Internal Ribbing of End Housings*

ROTOR COOLING OIL SCHEMATIC

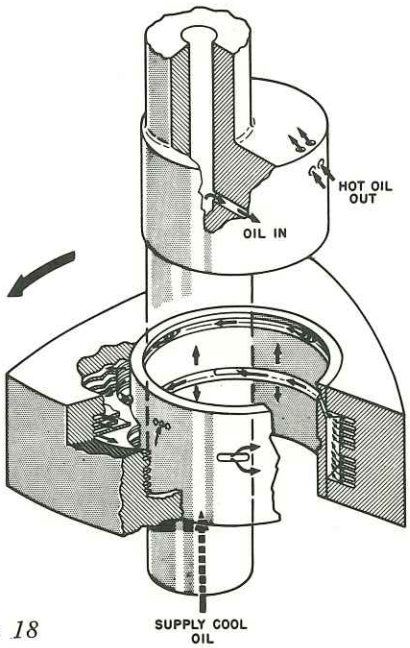



*Figure 18*

SEALING – The sealing system was orginally copied from the small engines. Whereas the bigger engine gave advantages in some respects; many of the sealing components were not durable enough and a more rugged system was worked out and incorporated in the rotor. This seal arrangement proved more durable and tighter.

TEST RESULTS – The early experimental engines were equipped with a side intake port as depicted in *Figures 5 through 13*. Typical full throttle performance data of this configuration are shown in *Figures 19 and 20*. Best power at 5500 RPM was 100 BHP equivalent to 1.67 hp per cu. in. displacement; it is not yet limited by the port areas and airflow. The torque and specific fuel consumption curves are relatively flat over the speed range of 2000 to 6000 RPM. At the lowest speeds, the torque drops somewhat, a shortcoming which was corrected by later designs.

*Figure 20* shows the volumetric, mechanical and indicated thermal efficiencies. Consistent with the torque characteristic, the volumetric efficiency was high over a wide speed range and reached, with a side port intake, values of 90% and more. The mechanical efficiency also reached almost 90% at the lowest speeds, indicating that the mechanical friction of the engine is low. The mechanical efficiency drops at higher speeds, but was improved by further development. It should be noted that at 100 hp, the mechanical efficiency still was approximately 80%. The indicated thermal efficiency for best economy reached 34% over a wide speed range.

*Figure 21* demonstrates the power and fuel consumption as a function of the fuel/air ratio for full throttle constant speed. The values of 0.061 fuel/air ratio for best economy and 0.073 for best power show good combustion characteristics in the engine.

*Figure 22* shows the performance of the same engine but equipped with a peripheral intake port. This port further reduced the intake losses resulting in correspondingly higher airflow and output. The engine delivered 124 hp

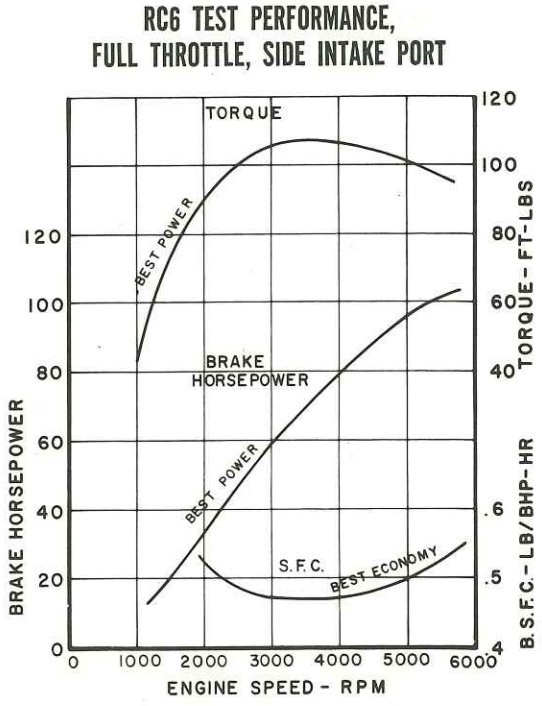



*Figure 19*

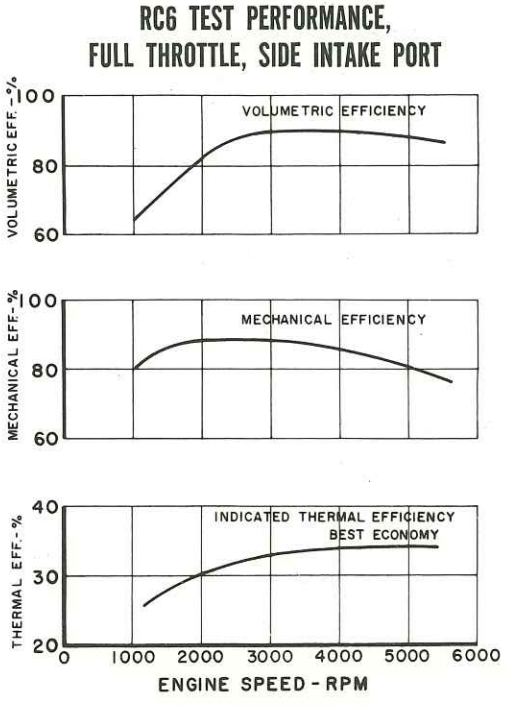



*Figure 20*

at 6500 RPM, equivalent to 2.05 hp per cu. in., and again showed promise of further output increase with speed and some development.

The volumetric efficiency and airflow in *Figure 23* show the high-speed characteristic of this engine, with volumetric efficiencies exceeding 100% and a steadily increasing airflow with speed reaching almost 1100-lbs/hr at 6500 RPM.

Supercharging of the engine with carburetion produced the expected result in output growth with manifold pressure, namely, some 30% with 6" Hg boost pressure before the carburetor, see *Figure 24*.

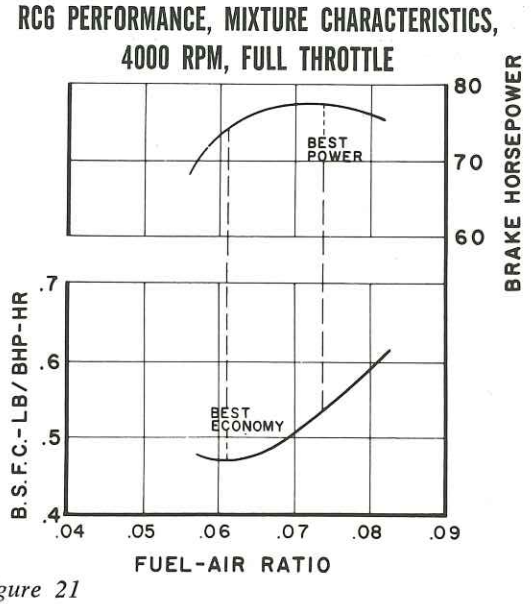



*Figure 21*

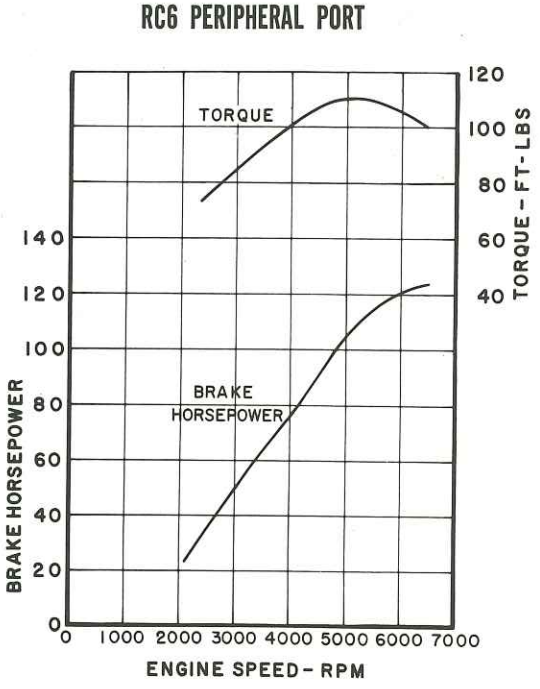



*Figure 22*

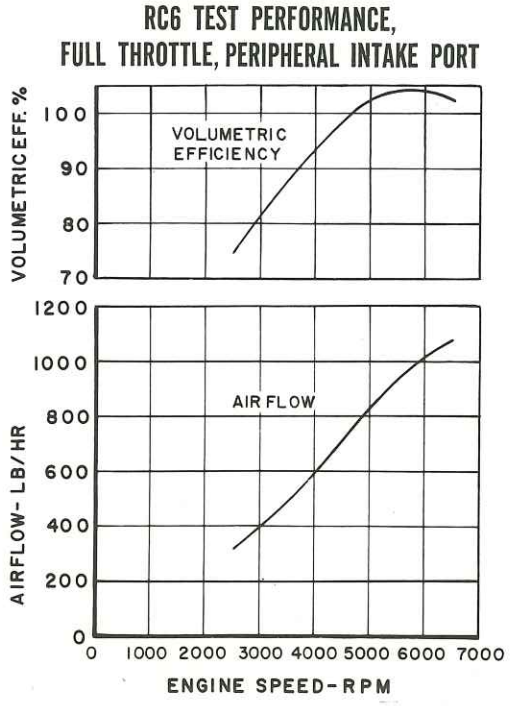



*Figure 23*

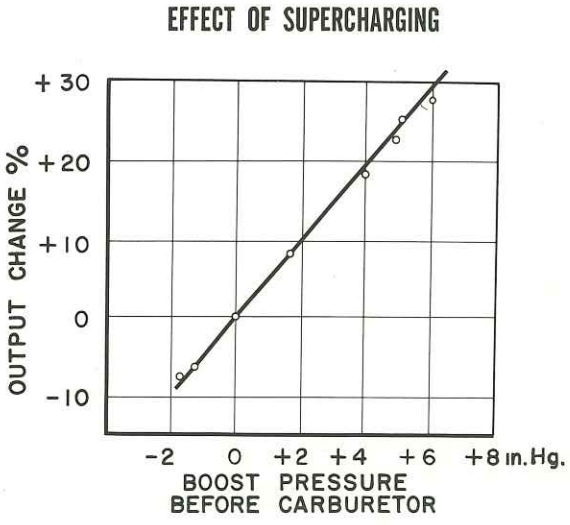



*Figure 24*

The peripheral port engine was slightly inferior with regard to air utilization as compared to the side port engine. This fact was reflected in a somewhat higher specific fuel consumption. The peripheral intake port configuration provides good breathing capacity and further development was subsequently concentrated on this basic design. Important areas under investigation were improved rotor housing cooling while still using the original basic cooling configuration as described and pictured above, a very much simplified and more effective cooling arrangement for the rotor, design and manufacturing refinements of the gas and oil seals, and better combustion.

The high speed capability of this engine series was improved by optimization of the size, shape, and timing of the intake and exhaust ports, and was demonstrated in various test runs. A single rotor engine produced a maximum output of 143.8 BHP at 6500 RPM with a BSFC of .54-lb/hp/hr. This power is equivalent to close to 2.4 BHP/cu. in. displacement. The highest BMEP at 5000 RPM was 163 PSI.

Mixture control curves run at speeds from 5000 to 7000 RPM indicate good combustion characteristics although the fuel/air ratio range for best power and best economy became somewhat narrower with increasing speed. *Figure 25* depicts the results of such a test series, namely, the output and specific air and fuel consumption in relationship to the fuel/air ratio for various constant speeds. This data exemplifies the transfer of the design parameters as described in reference (7) and below to higher speeds and apex velocities.

Similar optimization was applied to the medium and low speed range leading to improved performance over the initially reported data. A specific fuel consumption for best economy of .46-lb/hp/hr. was achieved.

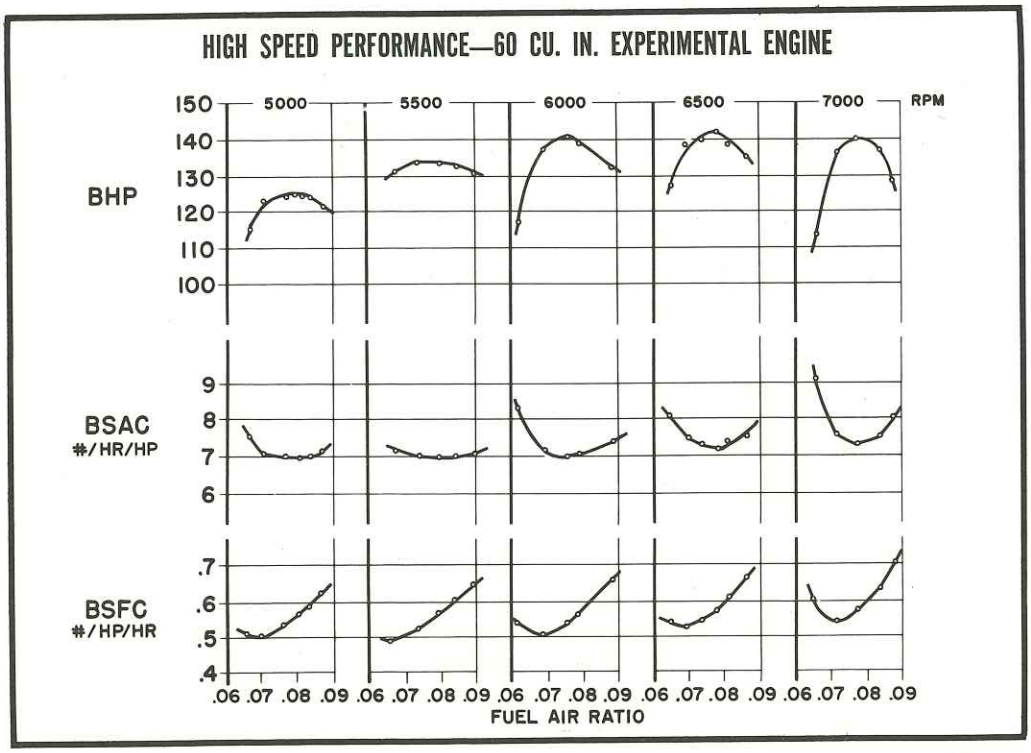



*Figure 25*

STARTING AND IDLING – The friction mean effective pressure at low speed which is important for the starter requirements, especially at low ambient temperatures, and for the idling characteristics, is qualitatively shown in *Figure 26*. Tests in this area have shown that the engine does not have a high break away torque such as reciprocating engines have.

The highest torque occurs at a moderately low speed where the mechanical friction is low but apparently the work done during compression is not fully recovered by the expansion due to some deficiency of the gas sealing at these low speeds. This loss disappears at slightly higher speeds which give the lowest friction mean effective pressure of the engine. At higher speeds, the expected rise occurs as a result of losses by mechanical and gas friction, oil churning, and heat dissipation.

Starting of the engine, down to temperatures below 0°F, was consistently accomplished with speeds in the order of 200 RPM. The starting speed in itself does not appear to be the only factor for starting the engine. Various indicator diagrams during the starting procedure have shown that a high acceleration results in firing in the first few cycles of the engine. The single rotor engine idles in the 350 RPM and runs at wide open throttle in the 400 RPM range.

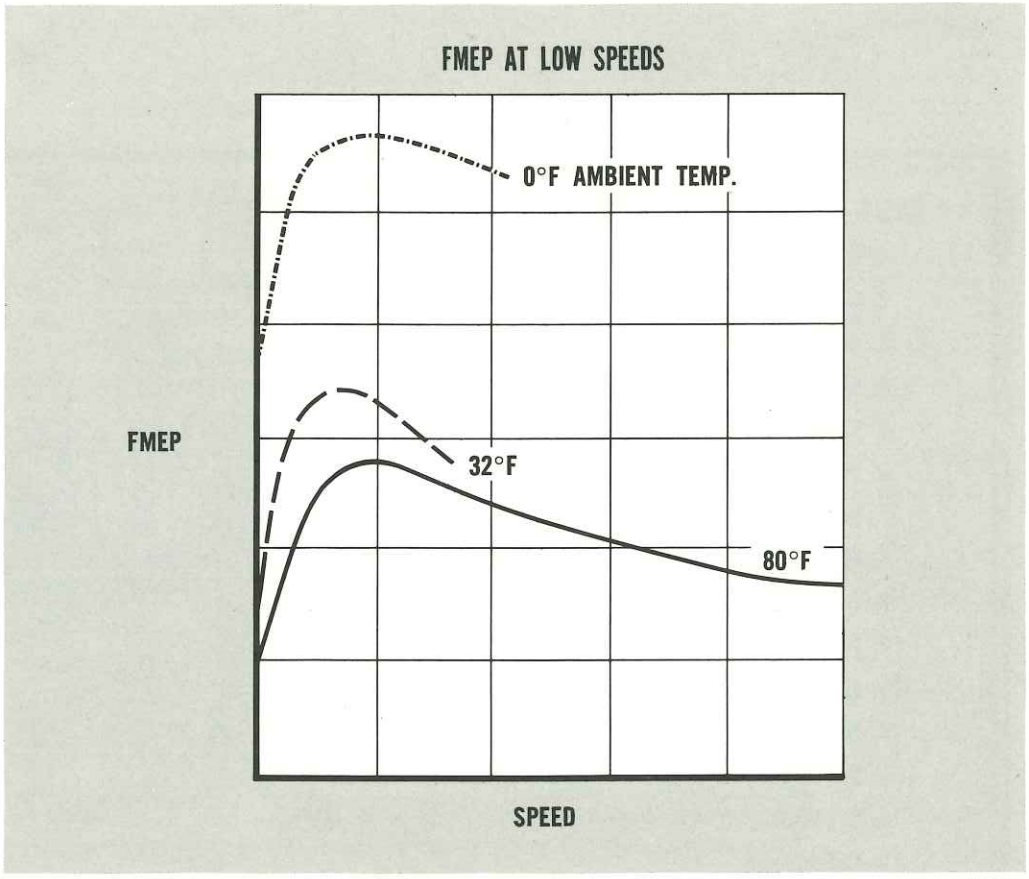



*Figure 26*

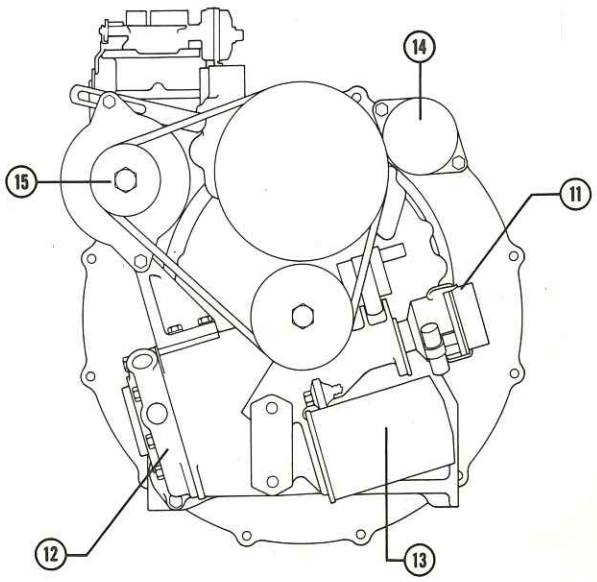
FUELS AND DURABILITY – A combination of optimizations with intake and exhaust timing and carburetor venturi sizes can be used to adjust the torque characteristic to specific requirements. Various motor fuels were also successfully run covering regular automotive gasoline, non-leaded high octane fuel, leaded fuels, and aviation gasoline grade 115/145.

Engine durability was repeatedly demonstrated by constant speed and load as well as cyclic test programs; a representative example of the former type is a run at a speed equivalent to 90% of rated speed and a BMEP of 100 PSI of 1,000 hours duration, which was completed without overhaul of rotor and seal components, and which exhibited excellent operating and wear characteristics of the engine components. Seal replacement on completion of the test was not indicated. Another remarkable achievement of this test is the fact that it required only one spark plug which still was operative after 1,000 hours, and which had accumulated a gap erosion of only .015".

Initial engine tests used automotive ignition systems and spark plugs of a normal automotive heat range. With increase in power and speed, it was found necessary to use a spark plug of very cold design for plug durability, and some plug fouling was then encountered. An auxiliary air gap at the spark plug reduced the fouling somewhat, but the periods between plug changes due to fouling and gap erosion were still not acceptable.

**SINGLE AND TWIN POWER PLANTS**

1. ROTOR
2. STATIONARY GEAR
3. ROTOR HOUSING
4. SIDE HOUSING – DRIVE SIDE
5. SIDE HOUSING – ANTI-DRIVE SIDE
6. INTERMEDIATE HOUSING
7. ACCESSORY HOUSING
8. MAIN BEARING
9. BALANCE WEIGHT
10. FLYWHEEL CUM BALANCE WEIGHT
11. IGNITION CONTACT MAKER
12. OIL PUMPS
13. OIL FILTER
14. STARTER
15. GENERATOR
16. WATER PUMP



A Curtiss-Wright designed condenser discharge system, which produced a voltage rise rate four times that of an automotive system, reduced gap erosion and sensitivity to fouling deposits considerably.

Another endurance run programmed for severe cyclic variations of load and speed with a maximum BMEP of 140 PSI at 5000 RPM over a period of 300 hours was also successfully completed. Oil consumption was comparable to reciprocating engines; the fuel did not contain any oil or lubricating additive.

**POWER PLANTS:** Based on the experimental engine, which had all accessories test stand driven, a single and twin power plant were designed. The longitudinal sections and rear view are shown in *Figure 27*. The front and rear section of the single and twin are identical (apart from the ports)[5], the second power section of the twin is a mirror image of the single, but the rotor and rotor housing are identical. The shaft and bearing arrangement is very much simplified; the flywheel is combined with its adjacent balance weight; and the outboard bearings are eliminated; either engine has only two main bearings.

The arrangement, size and weight of the accessories do not permit obtaining the full benefit of the compactness of the power sections, and efforts to "miniaturize" these appear to be worthwhile and are in progress. A slightly different design of a twin engine was built and tested. *Figure 28* shows the spark plug side of the assembled engine with accessories and *Figure 29* the test installation.

[5] These designs were still based on side intake ports; conversion to peripheral ports simplifies the arrangement considerably.

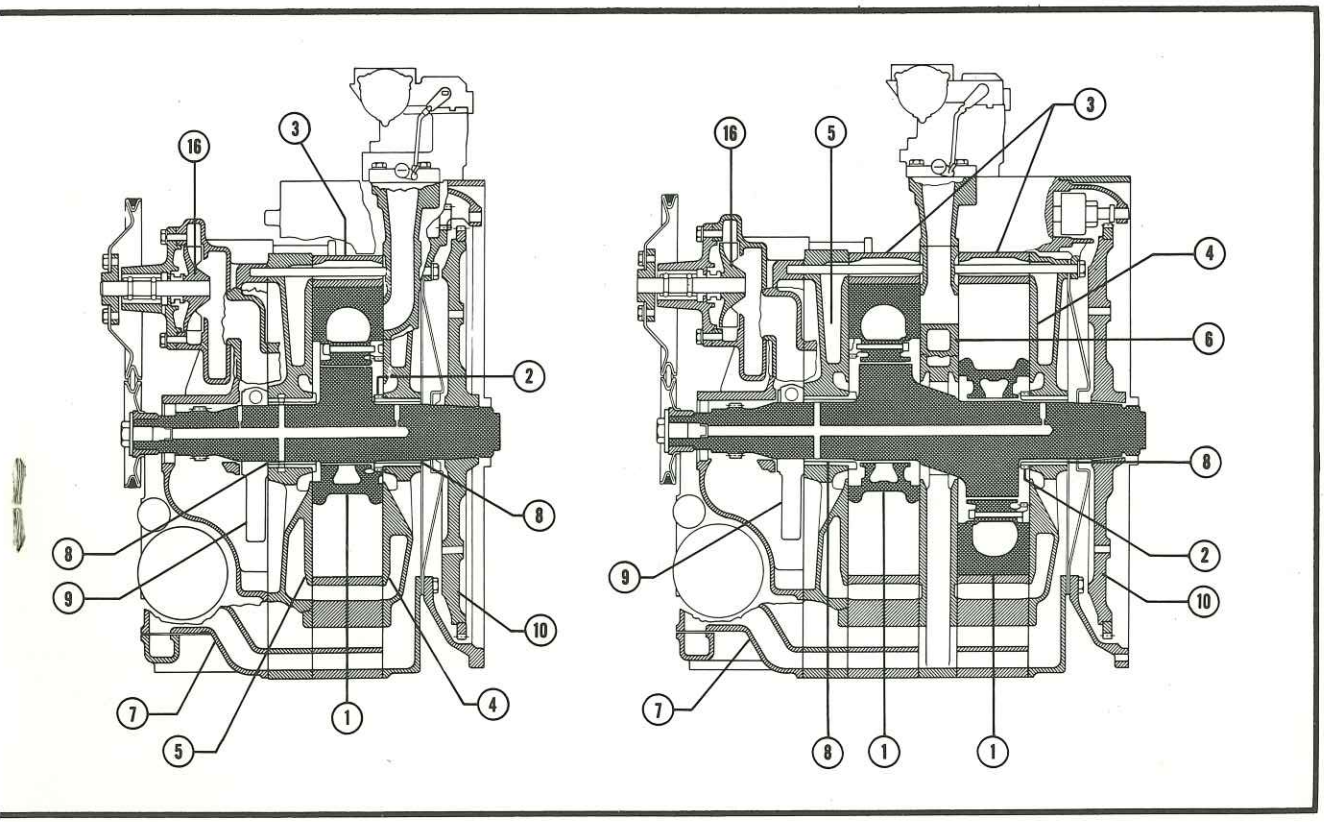



*Figure 27*

The engine essentially doubled the performance of the equivalent single rotor configuration; minor deviations in the torque characteristics were mainly influenced by the manifold arrangement. Initial performance with ports not yet optimized was 215 hp at 5000 RPM and a BSFC of .51-lb/hp/hr. The specific air and fuel consumption data of the twin engine are slightly better than that of the single. Applying the latest knowledge on porting, timing, and manifolding will result in a very compact and attractive power plant.

The instantaneous rotor torque of a twin is very much better than that of a single as is shown in *Figure 30* which depicts the torques for a 450 hp engine laid out as a single, twin, and four rotor configuration with geometrically similar power sections and equal apex velocity. Note that the torque is the result of gas forces only. The twin eliminates the negative torque and its torque peak is considerably lower than that of the single. The four rotor engine gives a very small torque variation. The mean torque decreases for the twin and four due to their smaller trochoid sizes and correspondingly higher rated speeds with the ratio of the square root of two.

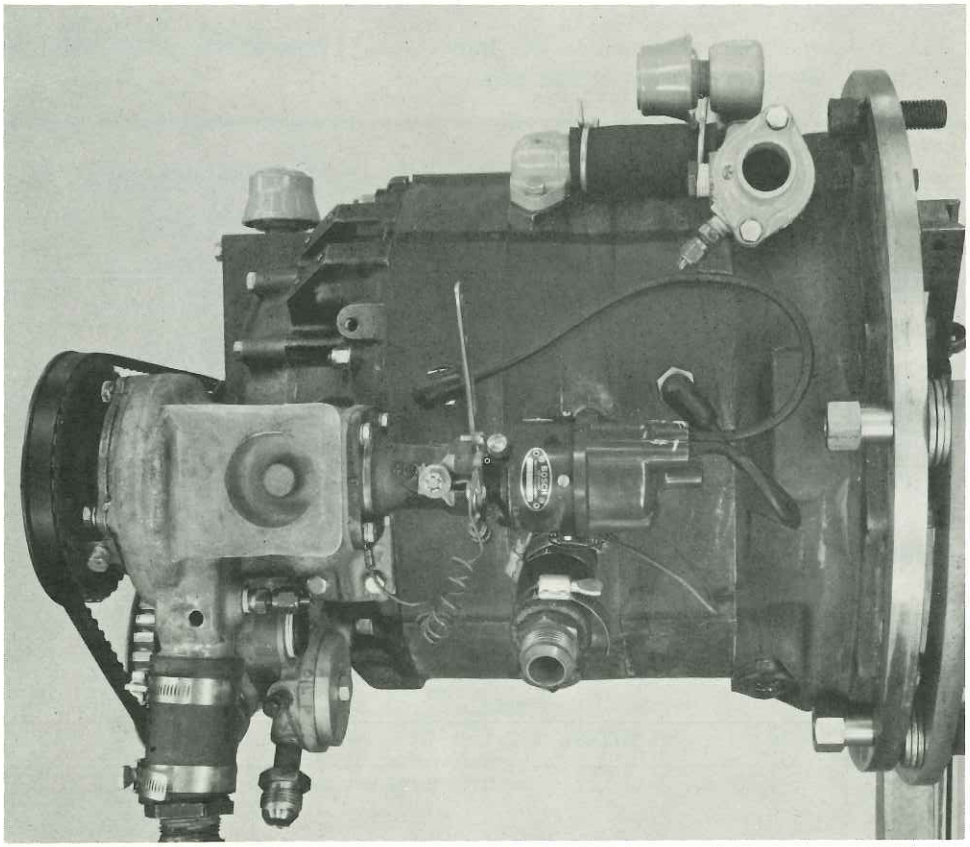


*Figure 28 — Photo of Twin Engine, Assembled*

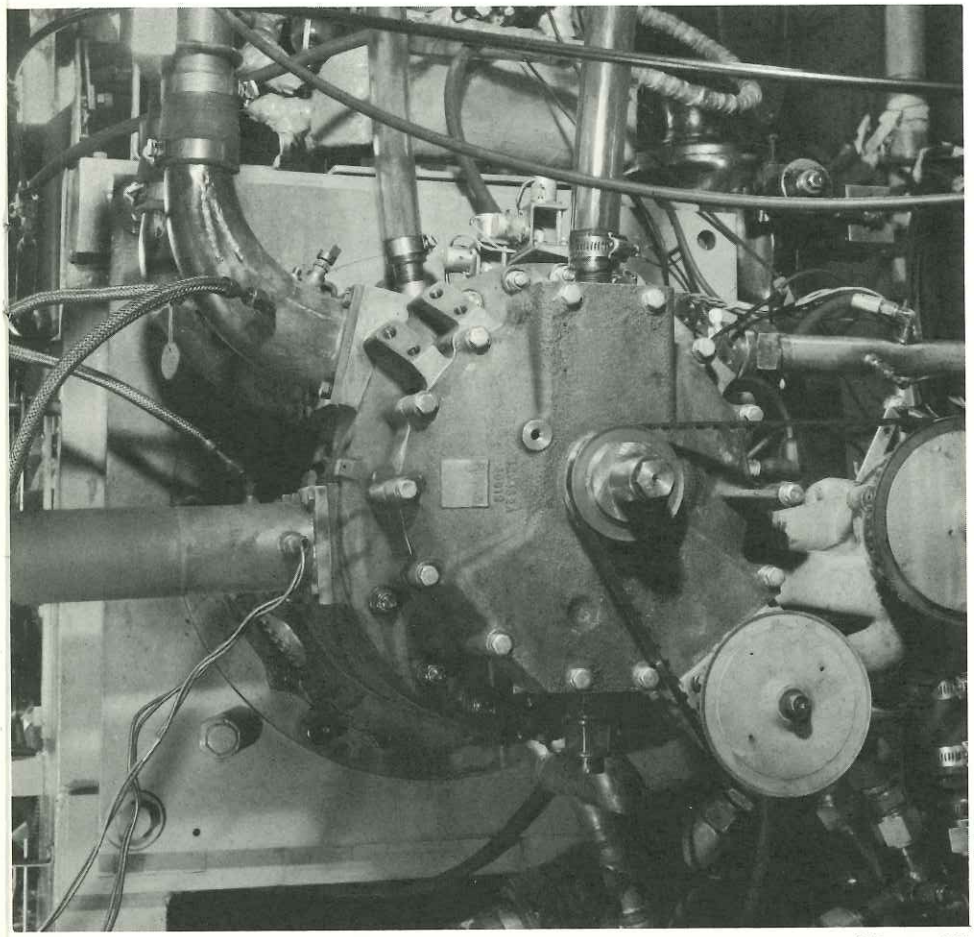


*Photo of Twin Engine on Test Stand*

*Figure 29*

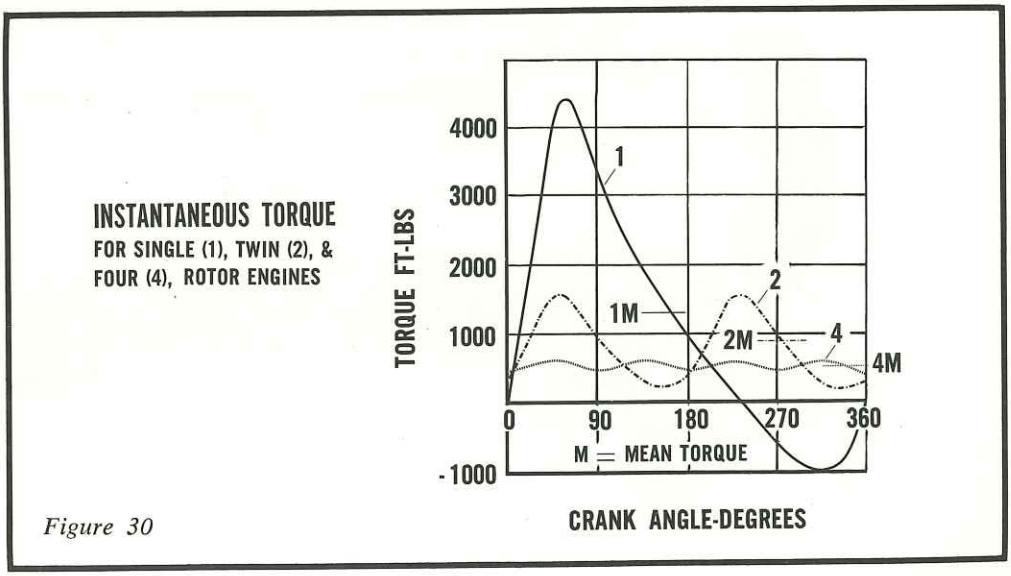



*Figure 30*

**FOUR ROTOR ENGINE 4RC6:** There are numerous possibilities for stacking power sections to a four rotor engine and after their careful evaluation, we selected the following configuration, again based on the 60 cu. in. components. In-line arrangement of the major trochoid axes, that means, in-line of the intake and exhaust ports, and the spark plugs, an integral one-piece crankshaft, uninterrupted sealing faces of the side housings, and parallel flow of the coolant and lubricant through the four power sections. The parallel flows had, apart from their advantages design-wise, the plus factor that all analytical and test results from the single unit could be readily applied to the four unit engine.

*Figure 31* pictures a side view of the engine which clearly illustrates some of these in-line design features. *Figures 32 and 33* show longitudinal and cross sections of the engine. The integral crankshaft, which runs in five main bearings, can be accommodated within the intermediate side housings by splitting these in a plane perpendicular to the engine axis and by splitting the bearings and stationary gears in a plane including the axis as illustrated. Assembly of the engine starts from the accessory gear box housing on the right to which the first main bearing with the inserted crankshaft is attached. Rotor and rotor housing follow, the rotor being slipped over the eccentrics of the shaft, and the addition of the side housing makes the power section complete. Rotor and side housings are doweled and clamped together the same way as on the engine in *Figures 5 through 7*.

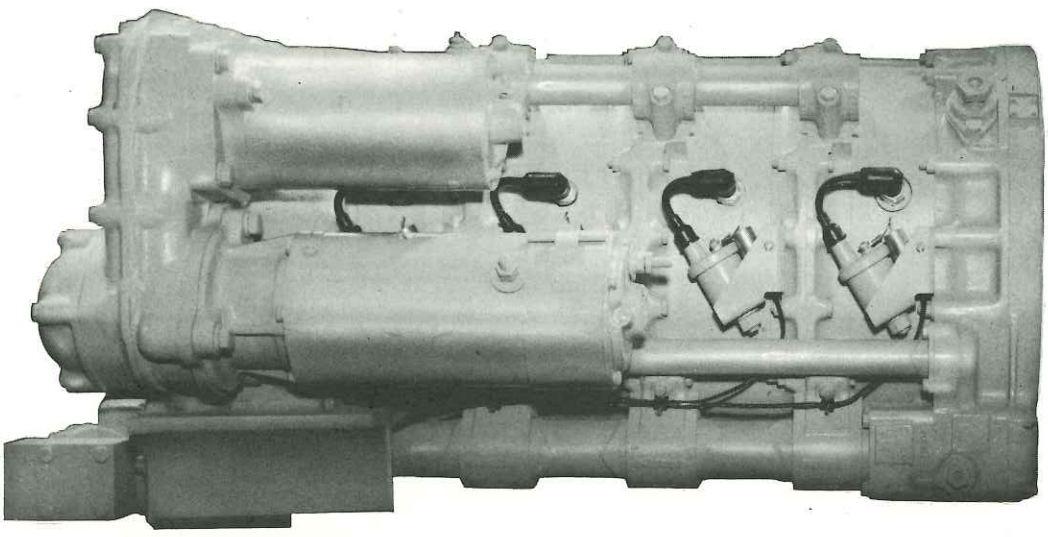


*Figure 31 — Side View of the Engine*

The second bearing and gear is assembled onto the shaft, bolted together, and piloted in the side housing. The split plane is serrated as a further precaution against dislocation of the two halves.

The side housing of the second power section is then piloted and bolted to the adjacent housing and the same sequence as before applies for this and the following power sections.

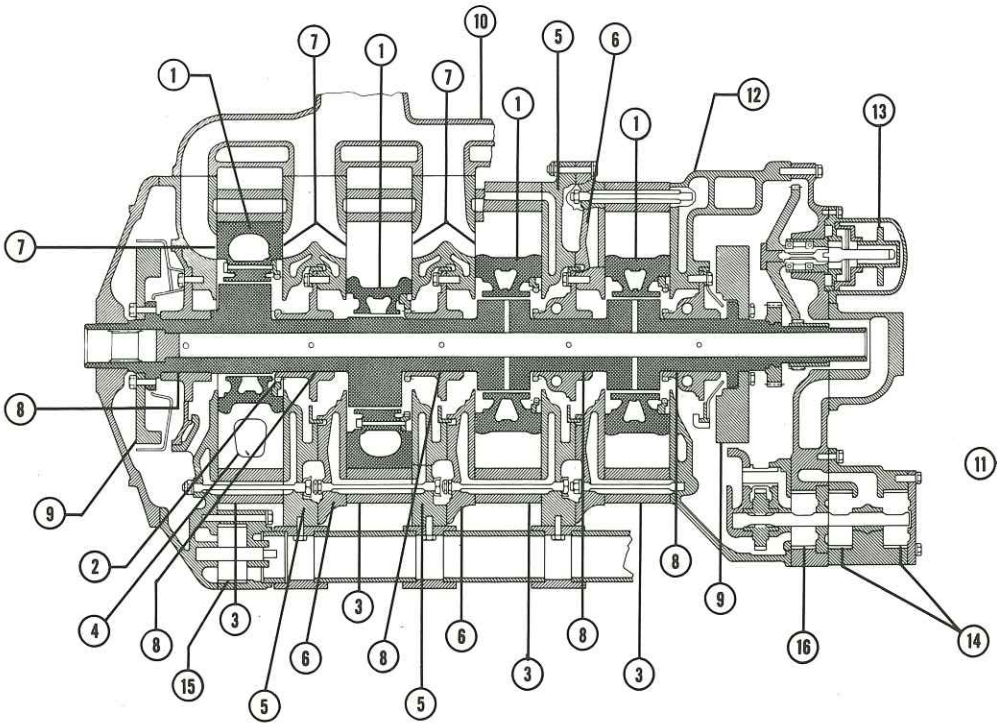



*Figure 32*

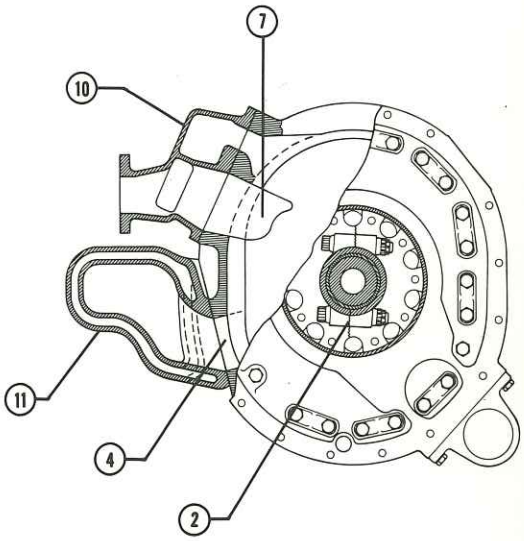



*Figure 33*

## 4RC6 ENGINE CROSS SECTIONS

1. ROTOR
2. SPLIT STATIONARY GEAR WITH MAIN BEARING
3. ROTOR HOUSING
4. EXHAUST PORT
5. SIDE HOUSING — DRIVE SIDE
6. SIDE HOUSING — ANTI-DRIVE SIDE
7. INTAKE PORT (DUAL INTAKE)
8. MAIN BEARING
9. FLYWHEEL CUM BALANCE WEIGHT
10. INTAKE MANIFOLD
11. EXHAUST MANIFOLD
12. ACCESSORY GEAR BOX HOUSING
13. IGNITION CONTACT MAKERS
14. OIL PRESSURE PUMPS
15. OIL SCAVENGE PUMP — FRONT
16. OIL SCAVENGE PUMP — REAR

This crank arrangement provides for power cycles at 90° intervals with the firing order of 1-4-2-3, but it leaves a small couple which has to be compensated by a balance weight at either end of the shaft. This penalty is, however, not serious since an additional torsional mass is probably desirable in any case, and the in-line arrangement provides for simple and light intake and exhaust manifolds, cooling lines, etc.

The crankshaft serves as the oil supply to the power sections. The return oil is delivered to the oil cooler and oil tank by two scavenge pumps located at the lower front and rear end of the engine.

The coolant is distributed by a tube along the housings, the exits are collected by the intake manifold which is thus heated for better fuel distribution at low ambient temperatures.

In order to achieve a good air and fuel distribution, two carburetors and dual side ports were employed. As illustrated by *Figure 31,* this engine was specifically designed for a low height and short length.

This was the first multi-rotor engine designed and built anywhere and we were ready for some surprises. We also got them but, fortunately, most of them were pleasant ones. Assembly of the engine did not pose any serious problem since all important components and their relative positions can be inspected during the assembly procedures in the same manner as on the single unit engine.

*Figures 34 and 35* show the engine rigidly mounted on the test stand with the engine accessories and the conventional instrumentation.

Only a few minor usual mishaps interrupted the first test series of this engine. Although the flow and torque variations, as shown before, are very small, observation of the smooth engine running confirmed this in an even more dramatic manner than figures can.

*Figure 36* depicts the full throttle performance of the first test engine, culminating in a maximum output of 425 hp at 6500 RPM, and showing relatively flat torque and fuel consumption curves, with optimum values at about 4000 RPM, namely, a torque of close to 400 ft. lbs. and a BSFC at best power of .51-lb/hp/hr. Application of the present state-of-the-art to this basic engine configuration would improve its output to some 550 hp at 6500 RPM.

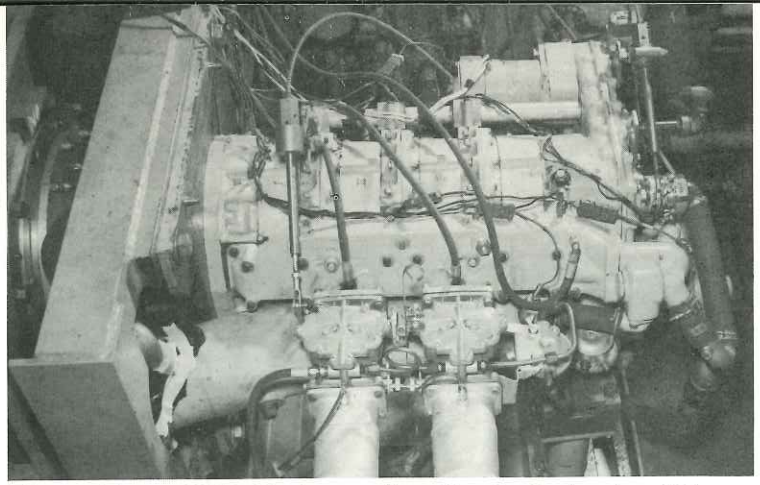


*Figure 34 — 4RC6 Engine on Test Stand—Carburetor Side*

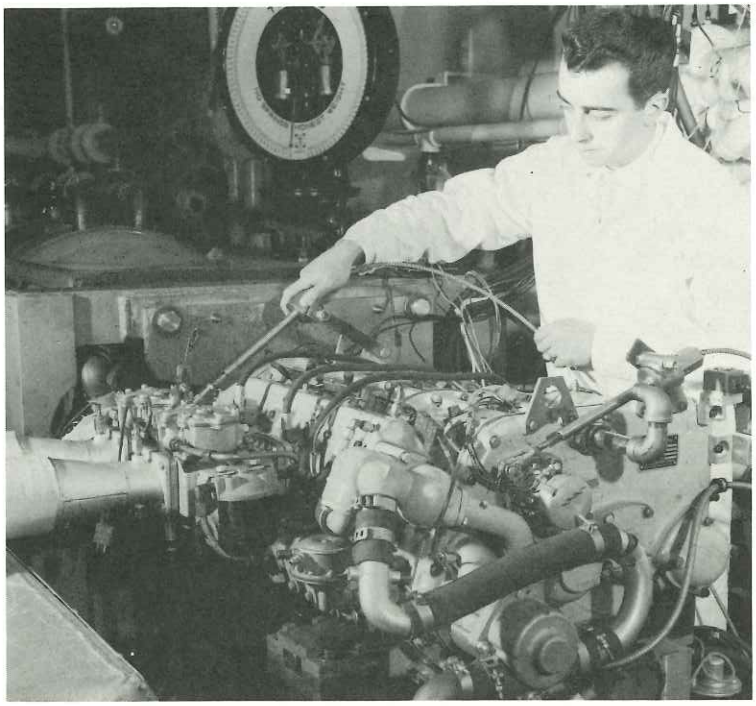


*Figure 35 — 4RC6 Engine —¾ Rear View Carburetor Side*

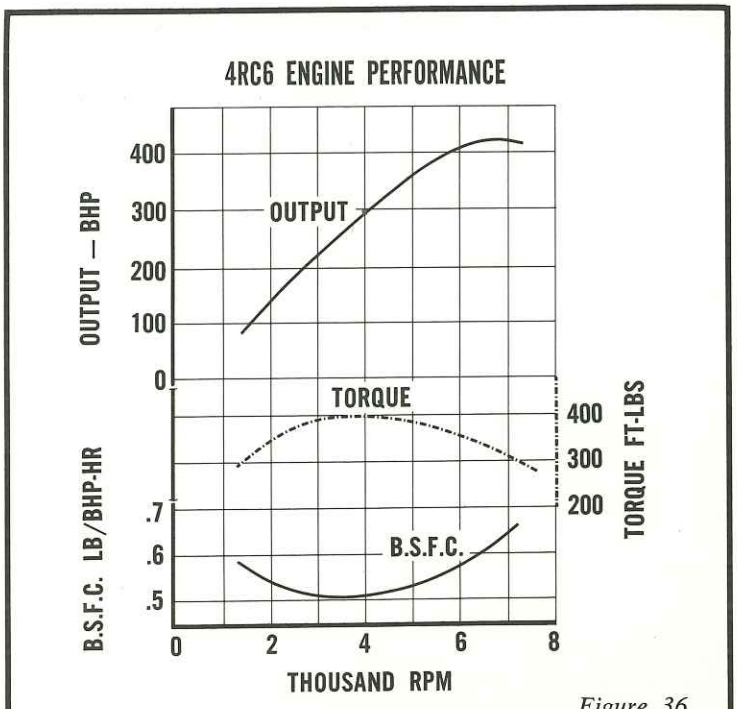



*Figure 36*

**SIZE EFFECTS:** In order to investigate the size effect on various engine phenomena, engines with very much larger and then with very much smaller displacements were investigated.

RC19 – First, the linear dimensions of the RC6 engine were literally blown up approximately three times to a some thirty times bigger displacement of 1920 cu. in., and such an engine, designated RC19, was built and tested. Only dimensions and features which affected stresses, heat transfer, and thermal gradients were adjusted. The engine configuration is essentially that shown in *Figures 5 through 7*. It had dual side ports each connected to a carburetor with manual control. *Figure 37* demonstrates the size of the power section.

This engine also started and ran at the first attempt and its operation was quite normal. Its breathing capacity proved very ample, even at high apex velocities, and the preliminary testing demonstrated the power potential of this engine size. However, it exhibited a phenomenon which did not occur on the smaller size engines with standard fuel, namely, detonation. It took place when standard fuel air mixtures and full throttle opening were approached. In this first test series, attractive power and fuel consumption figures could therefore not be realized and further investigation of this deficiency is required. It is too early to say that the size of the combustion chamber is the main cause of detonation. On the other hand, the engine helped to clarify other phenomena which were exaggerated due to the size and could more readily be investigated by instrumentation than on the smaller engine.

Table I gives the initial RC19 performance.

TABLE I

RC19 PRELIMINARY PERFORMANCE DATA

| | |
|---|---|
| POWER | 782 BHP @ 1525 RPM |
| BMEP | 106 PSI |
| VOLUMETRIC EFFICIENCY | 83 % |
| BSAC | 7.9 LB of AIR/HR/BHP |

*Figure 37 — RC19 Engine—Assembly of the Power Section*

SMALL AIRCOOLED ENGINE – Analytical studies and extensive experimental investigations led to the design parameters for the cooling of the engine housings with air. In order to substantiate these and other considerations related to size and speed effects, a small engine with some 4 cu. in. displacement was designed, built, and tested. Its housings are cooled by a centrifugal fan directly mounted on its crankshaft; the cooling fins and coolant flows are adjusted to the heat input to the rotor and side housings which varies along the trochoid perimeter. *Figure 38* shows the assembled engine from the spark plug side. The photograph illustrates the finning being compatible with the heat input and especially "heavy" in the combustion and exhaust port zones; it also indicates the cooling flows through the rotor and side housings.

The engine was designed for a maximum speed of 4000 RPM, a relatively low speed for its trochoid size corresponding to a mean apex velocity range which had not been explored so far.

The preliminary testing of two of these small engines confirmed the performance expectations; *Figure 39* gives the output of 1 to 3½ BHP at speeds from 1600 to 4000 RPM.

The engines can easily be started by hand with speeds in the order of 500 to 600 RPM; the initial idling speeds lie in the 1000 RPM range.

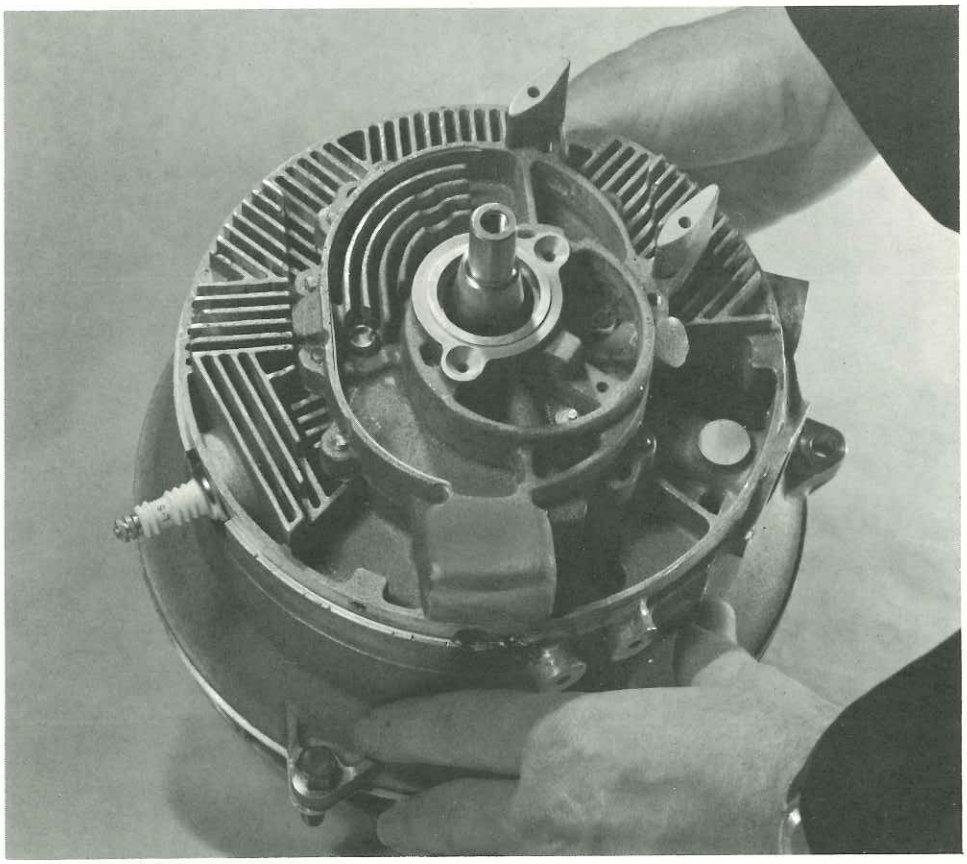


*Figure 38 — Photo of the Small Air Cooled Engine*

The inner wall temperatures of the rotor and side housings were well below the operating limit even with an ambient air temperature of 100°F. *Figure 40* depicts the rotor housing temperatures measured with thermocouples installed in the inner wall .1″ below the rubbing faces for hot day conditions and variable speeds and outputs. The temperatures reach a maximum at the spark plug location approaching 400°F at 4000 RPM; they slowly decrease from there in either peripheral direction. Reducing the power lowers the maximum temperatures and flattens the temperature profile along the trochoid perimeter.

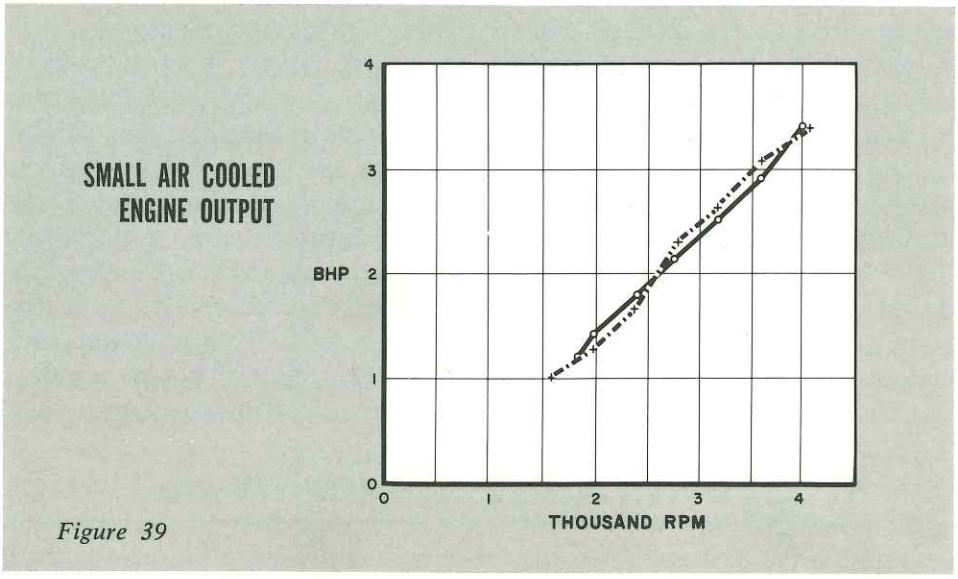



*Figure 39*

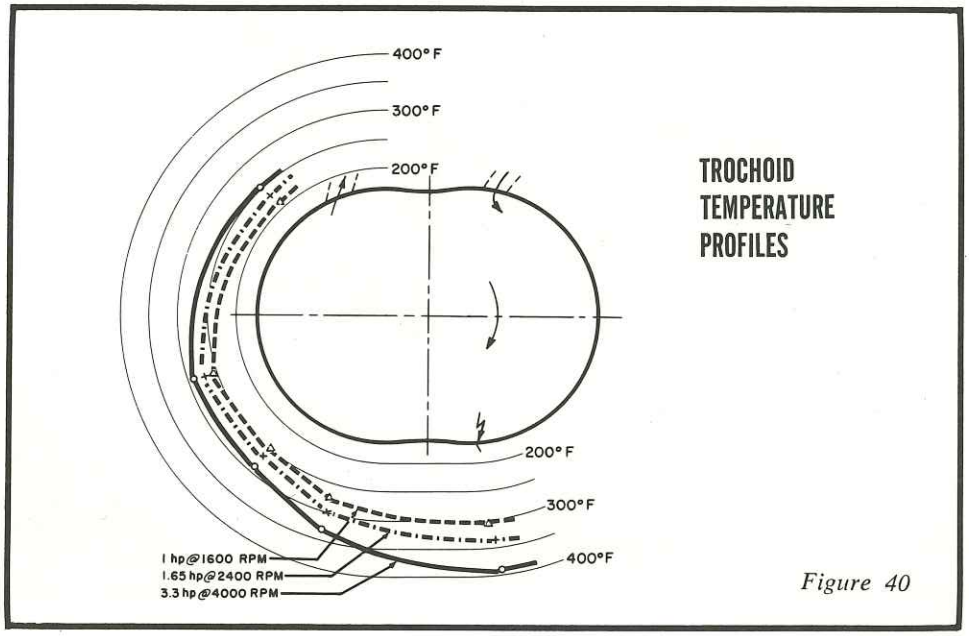



*Figure 40*

**SIZE EFFECTS ON ENGINE CHARACTERISTICS:** The small engine also confirmed the previous intuitive belief that the friction losses, as determined by motoring the engine, increase with decreasing engine displacement. This trend is even more pronounced if the FMEP is related to the same mean apex velocity instead of the actual crankshaft speed.

*Figure 41* illustrates qualitatively this tendency for engines of 60, 15 to 25, and 4 cu. in. displacement; the shaded areas indicate the spread of the test data as a result of different design and operating conditions. The rise in FMEP with decreasing displacement is very pronounced; it cannot be explained in all details as yet. Various factors contribute to this behavior, such as the ratios of chamber surface area to chamber volume, chamber volume to sealing length, rotor weight, bearing sliding velocities, etc. Quantitative analyses of these effects assist in the understanding of the friction loss relationships of the rotating combustion engines and in further improvements in this area.

If the FMEP of rotating combustion and reciprocating engines is related to mean apex velocity and piston speed respectively, their different characteristics become apparent. The FMEP of rotating combustion engines rises less steeply with speed than that of piston engines; it is, however, somewhat higher in the very low speed range. Seal

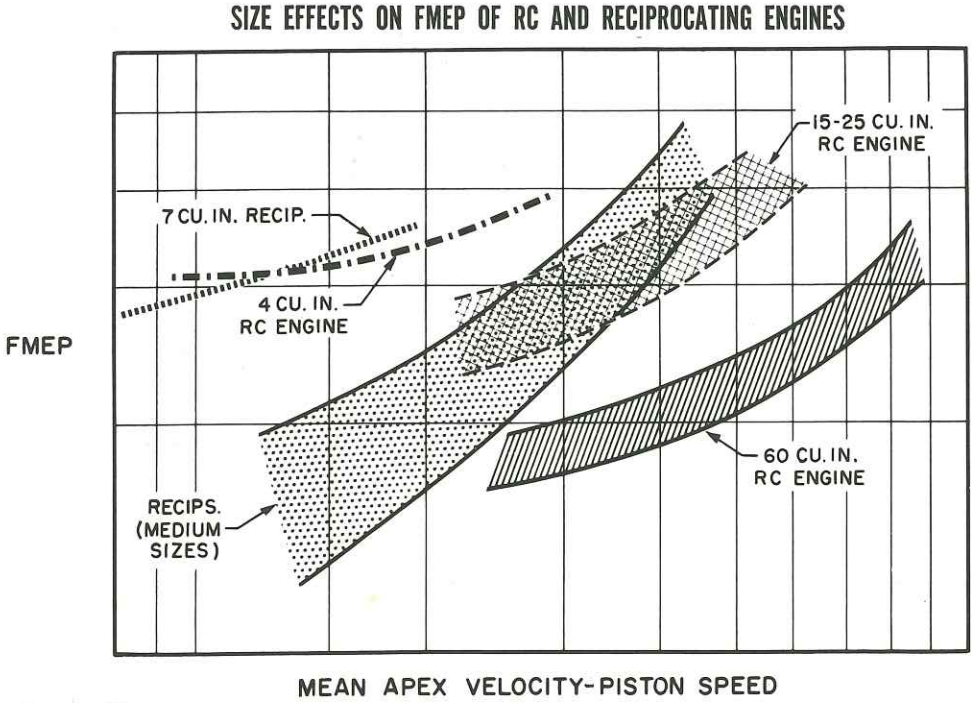

SIZE EFFECTS ON FMEP OF RC AND RECIPROCATING ENGINES
15-25 CU. IN. RC ENGINE
7 CU. IN. RECIP.
4 CU. IN. RC ENGINE
FMEP
60 CU. IN. RC ENGINE
RECIPS. (MEDIUM SIZES)
MEAN APEX VELOCITY-PISTON SPEED

*Figure 41*

operation may play a role in this respect as previously indicated by low speed tests. The FMEP of medium size rotating combustion engines is considerably less than that of medium size reciprocating engines in the operating and especially high speed range. This advantage diminishes with decreasing displacement; very small engines show approximately equal FMEP.

The mean apex velocity can serve as a valuable indicator for the prediction of engine performance. *Figure 42* illustrates this on engines of 60 and 4 cu. in. Although low apex velocities reduce the peak compression pressures, they still permit effective and economic engine operation as is shown by both the 4 and 60 cu. in. engines. In the medium and high speed range, this velocity effect is replaced by those of the porting, timing, and manifold layouts.

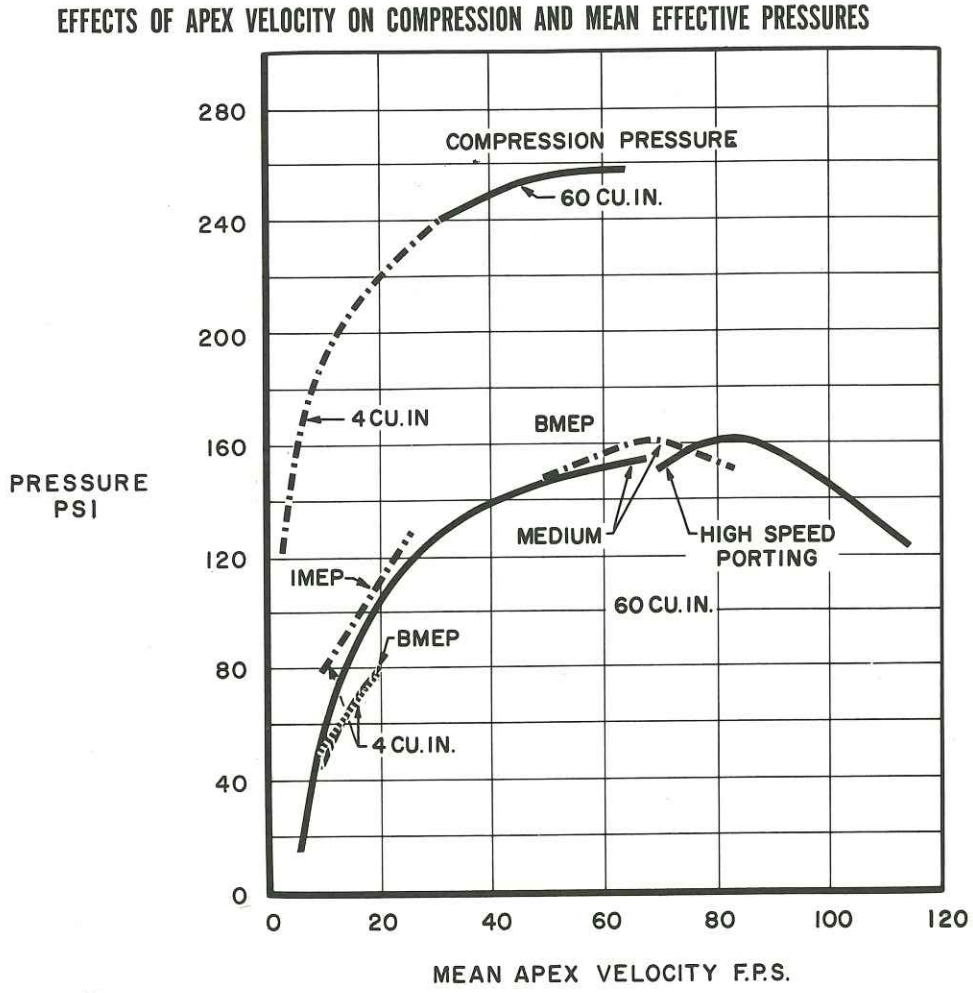



*Figure 42*

**DESIGN PARAMETERS:** Progress in the technical evalution and application of the rotating combustion engine has been made in a displacement range of 1:450, namely, from some 4 to 1900 cu. in., and a survey of the design parameters of this new prime mover is warranted. *Figure 43* represents an attempt to classify these parameters and to assess them in the form of relationships of brake mean effective pressure or overall efficiency versus speed or apex velocity.

By increasing the speed and gas velocities of a given size engine, the first problem encountered is combustion. Apart from the influence of the fuel type, combustion is affected by the manifold and porting arrangement, the combustion chamber shape, the transfer velocities inside the combustion chamber during the transition through top dead center, the compression ratio, and the spark plug location. We have developed an adequate combustion characteristic for current ratings and improved it also for higher speeds and velocities, i.e., higher power as demonstrated by the high speed performance, *Figure 25*. However, since part of the combustion chamber shape is solely determined by the geometry of the trochoid and the rotor envelope, the combustion phenomenon will present a challenge at the high speed and high power end.

Another factor will then limit power and economy with increasing speed, namely, friction caused by the friction of the rotor and its bearing, the churning of the scavenge oil within the power section, especially at the gear side, by losses due to the motion of the cooling oil inside the rotor, and by gas friction. Knowledge on all of these factors, both qualitatively and quantitatively, leads to design features which will reduce their detrimental effects. The influence of engine size and design was illustrated above in *Figure 41*.

At the low speed end, the engine performance is, apart from the porting, timing, and overlap of intake and exhaust, mainly influenced by the effectiveness of the gas seals. *Figures 26 and 42* indicate that seal operation is significantly affected by speed, and although improved seal designs helped this feature considerably, it appears that this limit is more severe than with conventional reciprocating engines.

An increase of the brake mean effective pressure, e.g., by supercharging, will raise the heat transfer rate to the rotor housing in its combustion section to such an extent that attainment of a wall temperature consistent with operational requirements is no longer assured. Of course, not very much can be done about the heat transfer on the combustion gas side but it has to be fully investigated in order to arrive

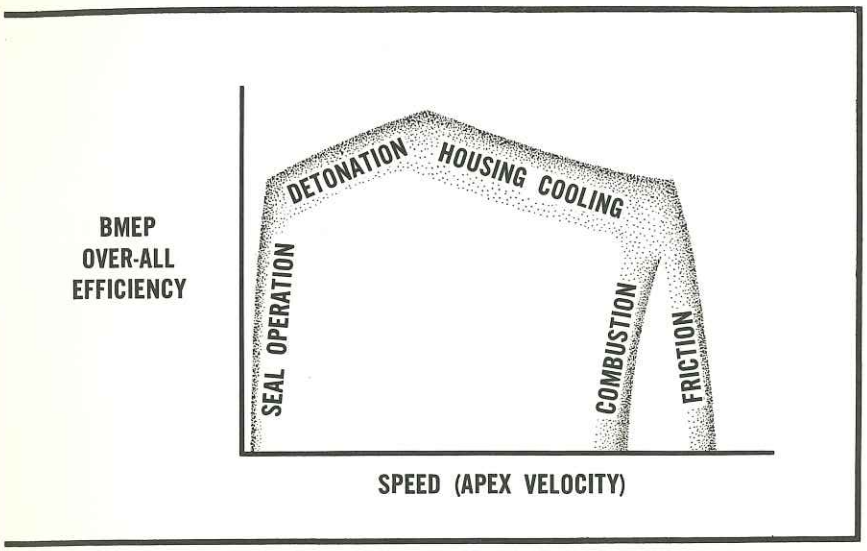



*Figure 43*

at an adequate cooling configuration on the coolant side. This heat transfer problem can only be partially resolved by analytical procedures, certain phases of which have to be supplemented by experimental testing. As a result of these evaluations, satisfactory relationships between the engine operating factors could be achieved.

On the coolant side, stationary and transient conditions have to be considered and the thermal properties of the housing material, namely, its conductivity, coefficient of expansion, diffusivity, and modulus of elasticity, the wall thickness of the housings, the coolant passage configuration, and the coolant medium play important roles and leave sufficient room to the ingenuity of the designer. Liquid coolants offer more margin in this respect than air, and early rotating combustion engines therefore employed water exclusively as a coolant. The small engines opened the field to aircooling; further investigations thereto are in progress.

Apart from the effect on friction, the rotor cooling does not appear to be a factor for the thermal limitation of the engine.

The last parameter is detonation which appears to reduce the permissible effective pressure at the low speed end, especially with large displacements. However, it has not been fully investigated as yet since it did not bother us with engines smaller than the RC19 or non-standard Otto-cycle fuels.

## BIBLIOGRAPHY


1. "Bauart und gegenwartiger Entwicklungsstand einer Trochoiden-Rotationskolbenmaschine" by F. Wankel and W. Froede, MTZ MOTORTECHNISCHE ZEITSCHRIFT, Volume 21, Edition 2/1960.
2. "Entwicklungsarbeiten an Dreh—und Kreiskolben-Verbrennungsmotoren" by W. Froede, VDI-ZEITSCHRIFT, Volume 102, 11 March 1960.
3. "Rotary Piston Engine Symposium," Anonymous, THE ENGINEER, February 26, 1960.
4. "The NSU-Wankel Rotary Engine" by R. F. Ansdale, AUTOMOBILE ENGINEER, May 1960.
5. "Konstruktion und Prufstandsergebnisse eines Kreiskolbenmotors KKM von NSU-Wankel," paper presented by W. Froede at the 8th International FISITA Congress, The Hague, May 1960.
6. "Rotating Combustion Engine," by M. Bentele, SAE TRANSACTIONS 1961, based on SAE Paper S-236 presented at a meeting of the Metropolitan Section of the SAE on March 10, 1960, and "Curtiss-Wright Rotating Combustion Engine Delivers 2 HP Per Cu. In.," SAE JOURNAL, June 1960.
7. "Curtiss-Wright's Developments on Rotating Combustion Engines," presented by M. Bentele at the International Congress and Exposition of the SAE in Detroit in January 1961, SAE Paper 288B, reprinted in SAE TRANSACTIONS 1961.
8. "Curtiss-Wright's Entwicklungen an Rotationsverbrennungsmotoren" by M. Bentele, MTZ MOTORTECHNISCHE ZEITSCHRIFT, Stuttgart, June 1961; Reprinted by TECHNISCHE RUNDSCHAU, Bern, Switzerland, August 11, 1961.
9. "Further Developments on Rotating Combustion Engines at Curtiss-Wright" by M. Bentele; Paper presented at a meeting of the Twin City Section of the SAE in Minneapolis, Minnesota, on March 21, 1962, SAE Paper S-348.

The information contained in this brochure is based on publications in the bibliography numbers 6 thru 9.

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